PROGRESS REPORT

FOR

NASA GRANT NAG-1-541

Vibration and Structureborne Noise

in Space Station

For the Period

January 1, 1986 - June 30, 1986

Prepared by:

Rimas Vaicaitis
Principal Investigator
Dept. of Civil Engineering
and Engineering Mechanics
Columbia University
New York, N.Y. 10027

(NASA-CR-177146) VIBRATION AND STRUCTUREBORNE NOISE IN SPACE STATION Progress Report, 1 Jan. - 30 Jun. 1986 (Columbia Univ.) 23 p CSCL 20A

N86-31336

Unclas
G3/71 43383

1.0 Summary of Major Activities

During the reporting period, the following major activities relating to the proposed work have been accomplished.

1.1 Conference papers presented.

1. Vaicaitis, R. and Bofilios, D.A., "Noise Transmission Into Enclosures," Modal Analysis Conference, Los Angeles, CA, Feb. 3-5, 1986.

2. Vaicaitis, R. and Bofilios, D.A., "Response Suppression in Composite Sandwich Shells," Vibration Damping Workshop II, Las Vegas, Nevada, March 1986.

1.2 Conference Papers prepared and accepted for presentation

1. Vaicaitis, R. and Bofilios, D.A., "Vibroacoustics for Space Station Applications," 10th Aeroacoustics Conference, AIAA, Seattle, WA, July 1986.

2. Vaicaitis, R., "Nonlinear Response - A Time Domain Approach," 10th Aeroacoustics Conferencee, AIAA, Seattle, WA, July 1986.

2.0 Technical Highlights

The technical background of structureborne noise generation has been described in the first two progress reports [1,2]. In what follows, a brief summary of new accomplishments during the reporting period is given.

2.1 Response of Stiffened Cylindrical Shells

Analytical models and computer programs for structural response calculations under action of mechanical point loads have been developed for single wall shells (composite or aluminum), double wall shells (composite or aluminum), and single wall or double wall circular plates (aluminum). These accomplishments have been described in the previous progress reports [1,2] and

Refs. 3-6. The design configuration of the habitability modules of the space station concept are expected to be discretely stiffened cylindrical shells with truncated cone type end caps or flat but stiffened circular end plates. The structural details of a typical (proposed) habitability module is shown in Fig. 1. Analytical formulations and response calculations have been performed for the case where the stiffened shell shown in Fig. 2 is represented by an orthotropic shell model. In this case the effect of rings and stiffeners is smeared into an equivalent skin. Then, the natural frequencies can be calculated by the procedures presented in Ref. 7. For the application to low frquency (below 1000 Hz) vibrations and noise generation, such a model might be adequate to evaluate vibration and noise transmission characteristics of space station habitability modules.

Natural Frequencies

The natural frequencies of an orthotropic shell are shown in Figs. 3-5 for several cases of different structural configurations. The structural parameters chosen are typical of the proposed habitability modules where L = 420 in, r = 78 in, h = 0.1 in, skin and all stiffening members are constructed from aluminum, and

A_r (cross-sectional area, ring) = 1.897 in^2

A_s (cross-sectional area, stringer) = 0.252 in^2

I_r (moment of inertia of ring about its centroid) = 5.294 in^4

I_s (moment of inertia of stiffener about its centroid) = 0.255 in^4

J_r (torsion constant for ring) = 0.1152 in^4

J_s = (torsion constant for stiffener) = 0.000302 in^4

$E = E_r = E_s$ (modulus of elasticity) = 10.0×10^6 psi

$G = G_r = G_s$ (shear moduli) = 3.846×10^6 psi

$\nu = \nu_r = \nu_s$ (Poisson's ratio) = 0.3

$\rho_s = \rho_r = \rho_s$ = 0.000259 $lb\text{-}sec^2/in^4$

The results shown in Figs. 3-5 indicate the effect of stiffening from increased number of frames. The modal frequencies for the first few circumferential modes (N = 0,1,2,3) are not affected much by the number of ring frames. However, for mode numbers larger than N = 3, the ring frames have a strong effect on modal frequencies. Furthermore, for the circumferential wave numbers larger than six and longitudinal modes higher than ten, the modal frequencies tend to converge to a single line. From these results, it can be seen that at each selected frequency several modes could be contributing to structural response.

Structural Response

The response of a stiffened shell shown in Fig. 2 was calculated for a variety of structural configurations. The inputs are point loads acting at $x = L/2$, $\theta_1 = 90^0$, $\theta_2 = -90^0$. These inputs are assumed to be stationary Gaussian white noise random processes characterized by truncated spectral densities

$$S_{F_1} = S_{F_2} = \begin{cases} 0.01 \ lb^2/Hz & 0 < f < 1000 \text{ Hz} \\ 0 & \text{otherwise} \end{cases} \tag{1}$$

The root mean square value of each force is about 3.33 lbs.

The response levels of a shell stiffened with 10 frames and stringers which are spaced 10 inches apart is shown in Fig. 6 for two values of shell skin thickness. These results were obtained from

$$RL = 10 \log \left(\frac{S_w \Delta\omega}{h^2}\right) \tag{2}$$

where S_w is the response spectral density, $\Delta\omega$ is the frequency bandwidth and h is the shell skin thickness. Since the spectral density is normalized with respect to the shell skin thickness, the RL represents relative values of the shell response. Damping was taken to be constant for all modes and equal to 0.02 (% of critical damping). As can be seen from these results, the response levels of a thinner shell are significantly higher.

2.2 Spound Pressure Levels

Parametric studies now are being performed to assess interior noise environment inside a habitability module to mechanically induced vibrations. The spectral densities of the input forces are taken to be those given in Eq. 1. It is assumed that some absorbent acoustic material is intact at the interior walls of the cylindrical enclosure.

The generated interior SPL in the enclosure at $x = L/2$, $\theta = 45^0$ and $r = 68$ in are presented in Fig. 7 for several geometric stiffening configurations. For the stiffened cases it is assumed that only transverse rings are present as stiffening

elements. As can be seen from these results, at low frequencies (below 150 Hz) interior noise levels for unstiffend shells are higher when compared to the results of a stiffened case. However, for frquencies above 150 Hz, higher noise levels might be generated at some frequencies for a shell stiffened with rings. This could be due to the fact that the acoustic modes at these frequencies are strongly coupled to the shell structural modes. The sound pressure levels for a pressurized shell stiffened with 10 rings and stringers which are spaced at 10 inches apart are given in Fig. 8. The results tend to indicate that more interior noise is generated by a thinner shell. The effect of stringers on interior noise levels is illustrated in Fig. 9. As can be seen from these results, the sound pressure levels do not change by much when stringers are added to the skin. The effect of locating the input point loads at different positions is shown in Fig. 10. These results correspond to a shell stiffened with ten heavy frames and stringers which are spaced 10 inches apart. The sensitivity of noise generation due to different locations of input forces is clearly demonstrated in this figure. Structural modes which are not excited for a particular forcing condition might become efficient sound radiators for a different set-up of input forces.

Numerical results were also obtained for a shell stiffened with relatively small ring frames. The structural parameters selected are:

$$A_r = 0.228 \text{ in}^2$$

$$A_s = 0.295 \text{ in}^2$$

$$I_r = 0.4437 \text{ in}^4$$

$$I_s = 0.5087 \text{ in}^4$$

$$J_r = 0.000122 \text{ in}^4$$

$$J_s = 0.000246 \text{ in}^4$$

The material properties are the same as given in the previous examples. A comparison of interior sound pressure levels for the two cases of different ring frame stiffening is shown in Fig. 11. As can be seen from these results, interior noise levels are significantly higher for a shell stiffened with small frames (frequency range 50-500 Hz). The results shown in Fig. 12 indicate the effect of increasing the number of small frames from 10 to 41. In the frequency ranges 50-200 Hz, 400-500 Hz, interior noise leves are higher for a shell stiffened with 41 small ring frames than for a shell stiffened with 10 heavy frames. However, for frequencies above 750 Hz more noise is transmitted when ring frames are large.

The results presented indicate the sensitivity of interior noise environment inside the habitability modules due to changes in structural, geometric and loading conditions. These results were obtained for a particular level of the point load intensity. At the present time, the magnitude characteristics and location of mechanical inputs that will be present in the habitability modules during orbital operations are not known. The measurements obtained for the Skylab operations shown in

Figs. 13 and 14 indicate typical interior noise levels. In general noise levels generated by various mechanical or electrical components are relatively low, but the total level might reach about 80 dB at some frequencies.

3.0 Future Work

We expect to continue the parametric studies of noise generation inside the habitability modules during orbital operations. Special emphasis will be placed on more realistic simulation of inputs resulting from mechanical and electrical devices. Furthermore, these analytical models will be extended to partitioned acoustic enclosures. In addition, we expect to conduct experiments on structureborne noise generation, propagation and transmission. These experiments will be performed utilizing the fuselage of a twin engine AeroCommander aircraft.

4.0 References

1. Vaicaitis, R., "Vibrations and Structureborne Noise in Space Station," Progress Report for NASA Grant NAG-1-541 for the period Jan 1, 85 - June 30, 85.

2. Vaicaitis, R. "Vibrations and Structureborne Noise in Space Station," Progress Report for NASA Grant NAG-1-541 for the period July 1, 85 - Dec. 31, 85.

3. Vaicaitis, R. and Bofilios, D.A., "Noise Transmission of Double Wall Composite Shells," 26th AIAA/ASME/ASCE/AHS SDM Conference Paper, No. 85-0604-CP, Orlando, FL, April 1985.

4. Vaicaitis, R. and Bofilios, D.A., "Noise Transmission of Double Wall Composite Shells," Fluid Structure Interaction and Aerodynamic Damping, ASME, Sept. 1985.

5. Vaicaitis, R. and Bofilios,D.A., "Response Suppression in Composite Sandwich Shells," Vibration Damping Workshop II, Las Vegas, Nevada, March 1986.

6. Vaicaitis, R. and Bofilios, D.A., "Vibroacoustics for Space Station Applications," 10th Aeroacoustics Conference, AIAA, Seattle, WA, July 1986.

7. Mikulas, Jr., M.M. and McElman, J.A., "On Free Vibrations of Eccentrically Stiffened Cylindrical Shells and Flat Plates," NASA TN D-3010.

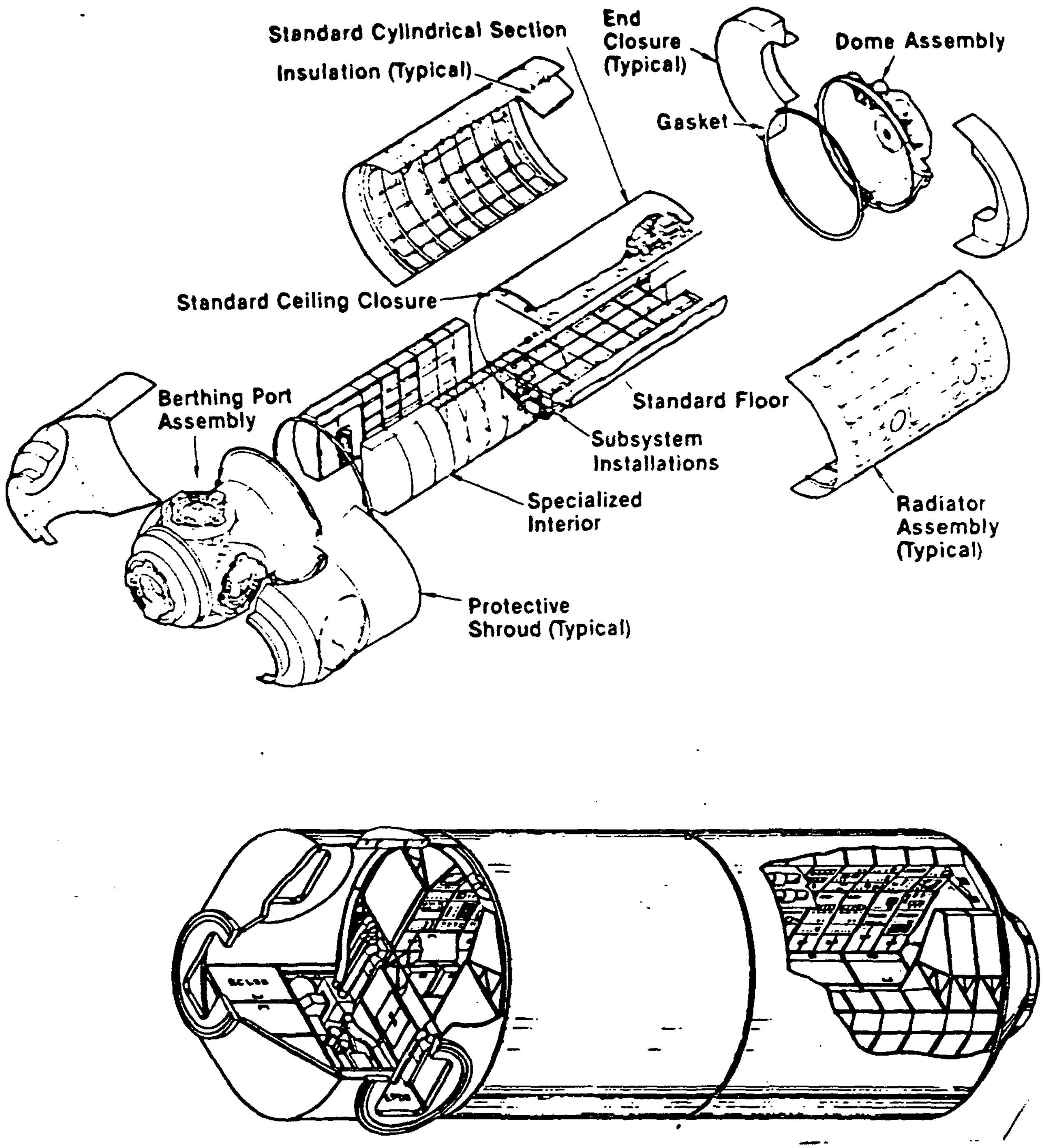

Fig. 1 Proposed geometry of habitability modules

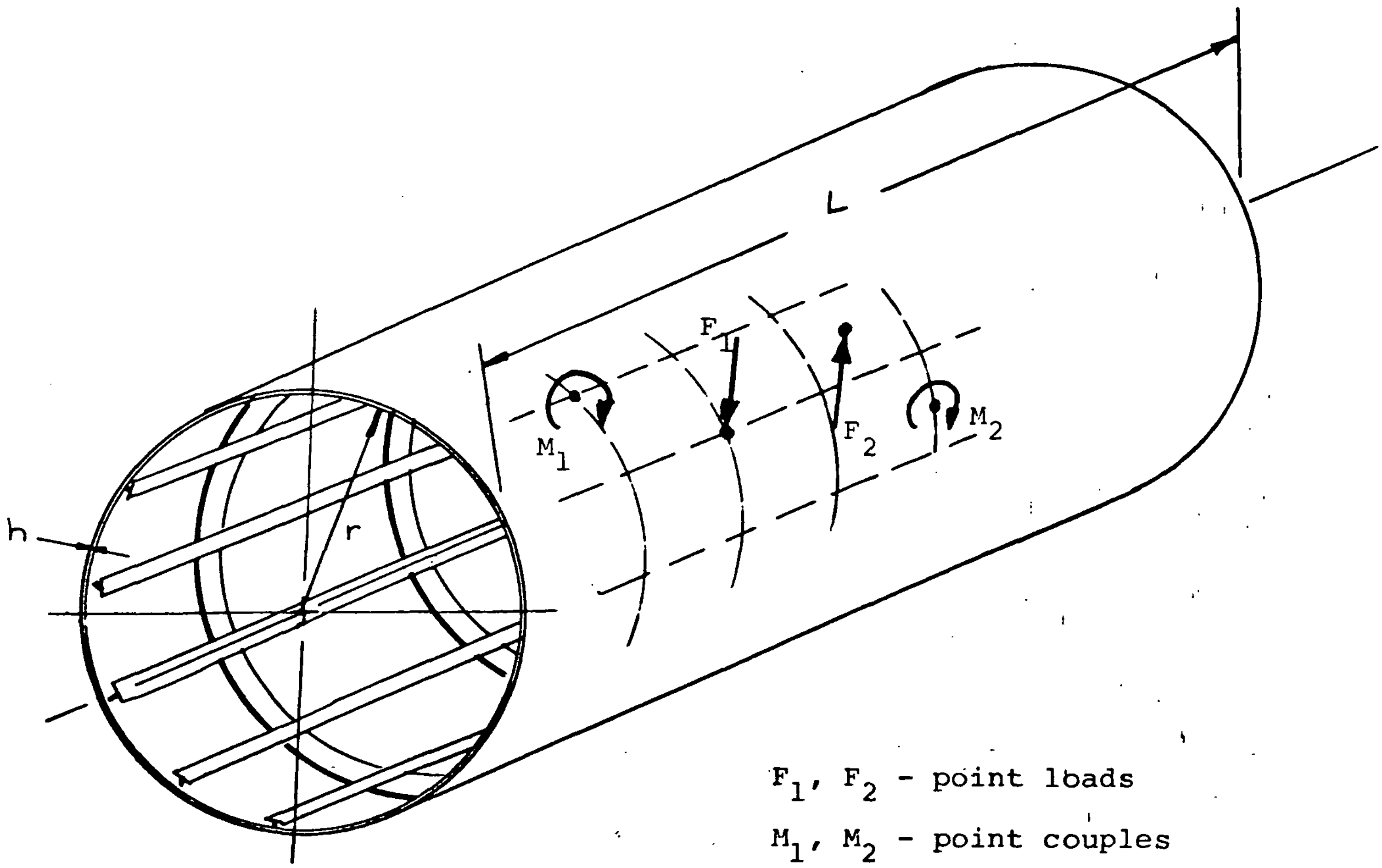

Fig. 2 Geometry of a stiffened cylindrical shell

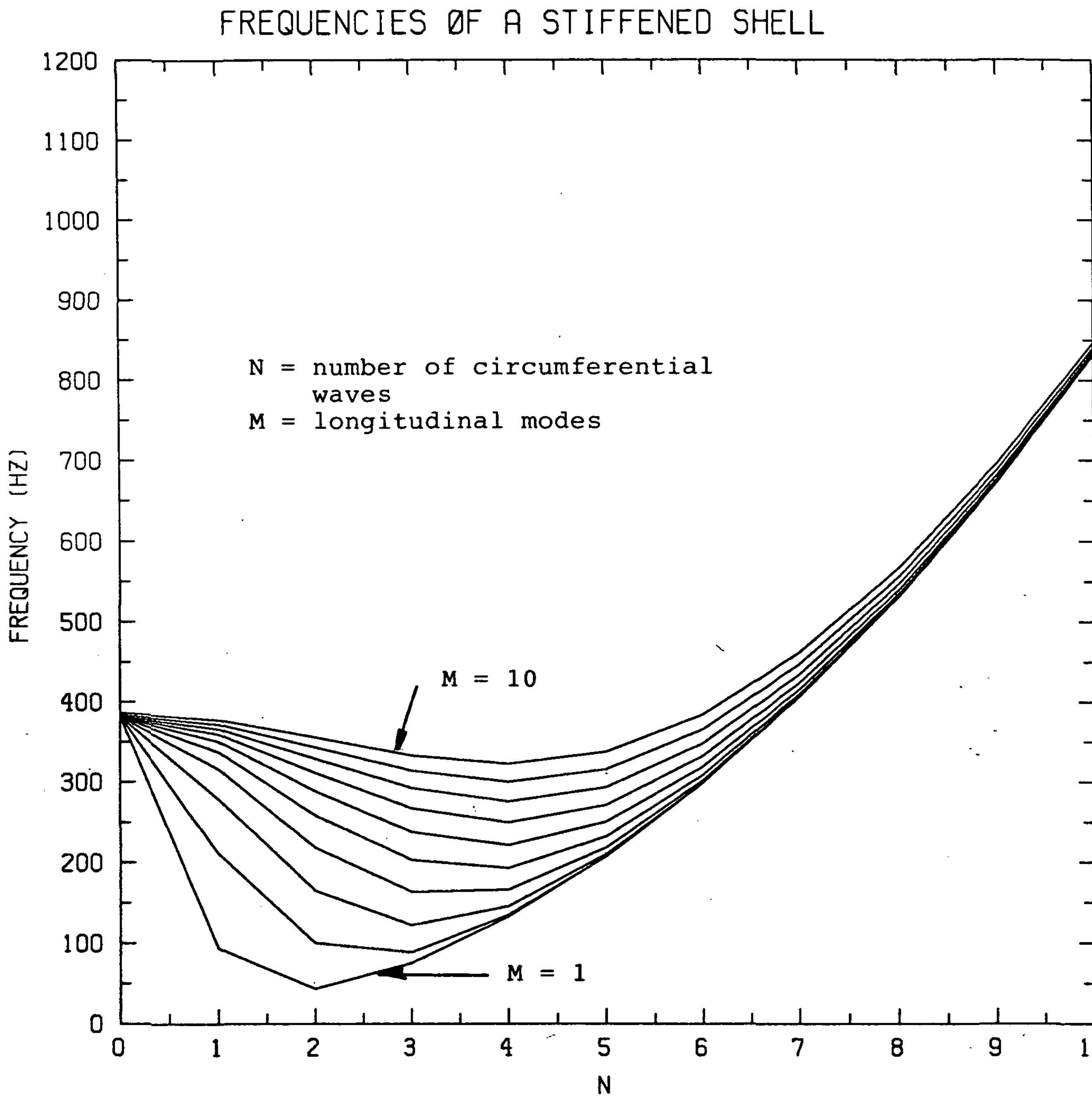

Fig. 3 Natural frequencies of a shell stiffened with 2 frames (distance between stringers = 20 in)

Frame : 1
File : USERDISK1:[LYRINTZIS]FREQ.OUT;108
Printed : 25-FEB-86 15:08:42

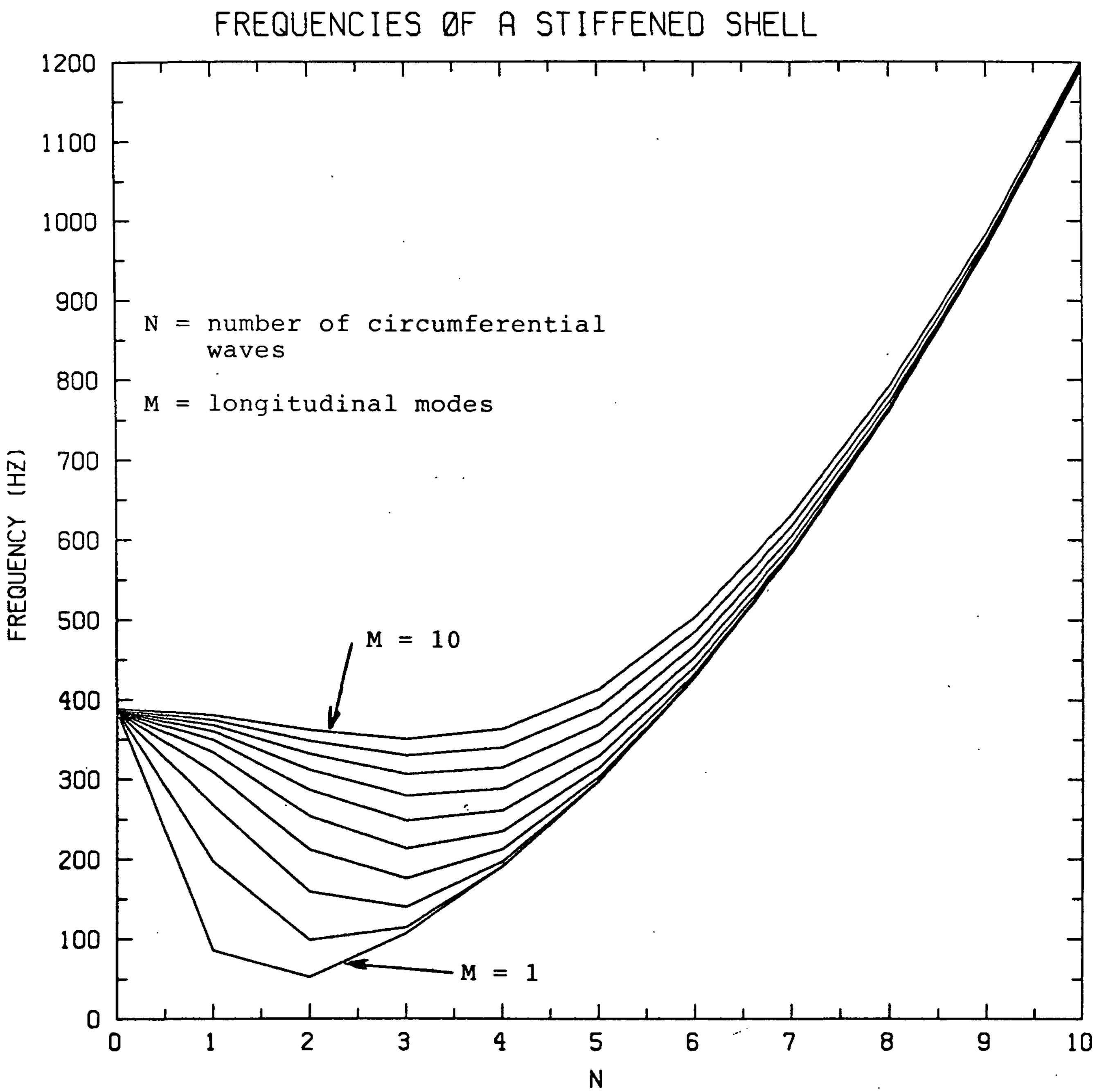

Fig. 4 Natural frequencies of a shell stiffened with eight frames (distance between the stringers = 20 in)

Frame : 1
File : USERDISK1:[LYRINTZIS]FREQ.OUT;111
Printed : 25-FEB-86 15:27:05

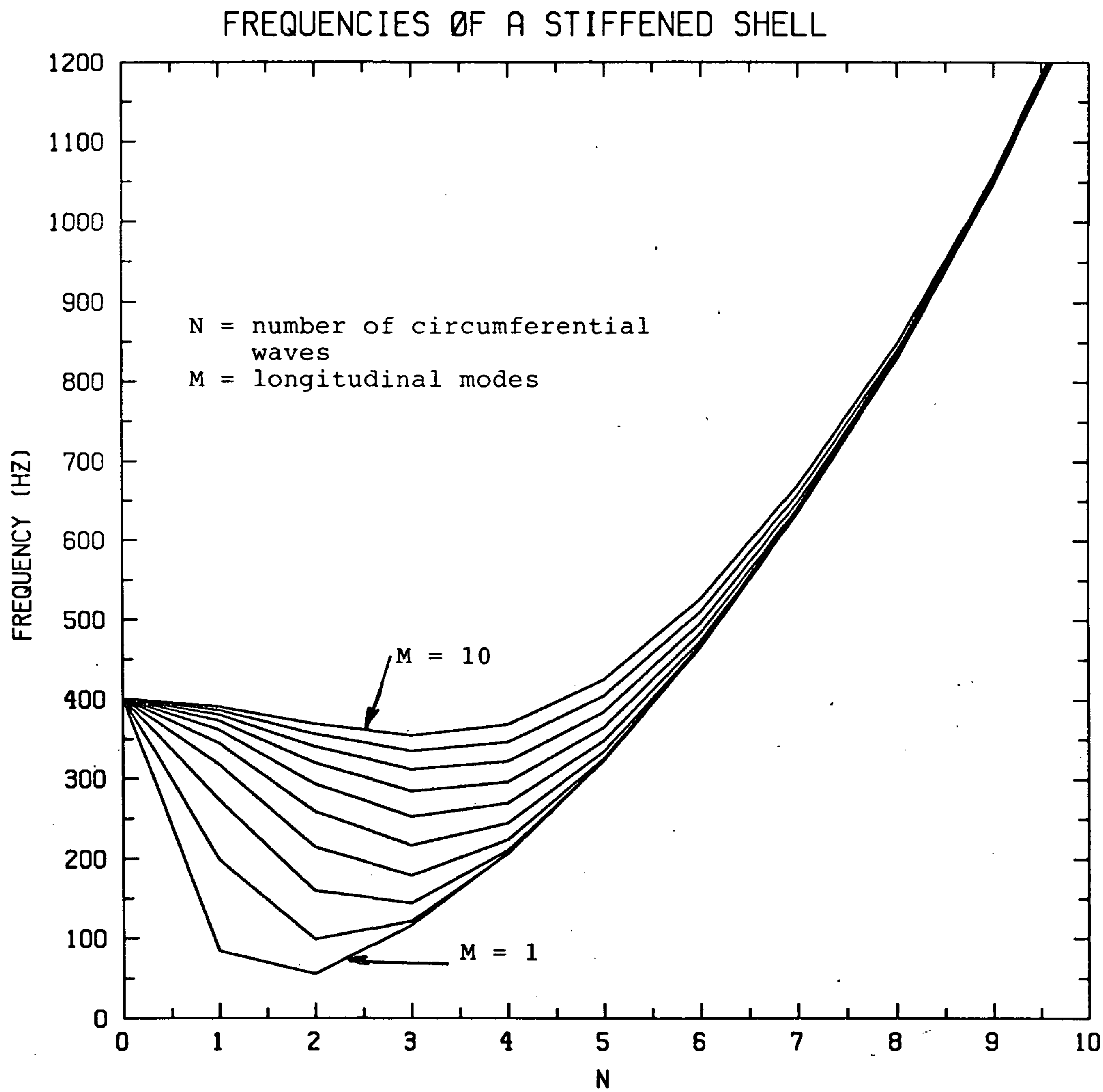

Fig. 5 Natural frequencies of a shell stiffened with ten frames (distance between the stringers = 20 in)

Frame : 1
File : USERDISK1:[LYRINTZIS]FREQ.OUT;84
Printed : 21-FEB-86 13:49:22

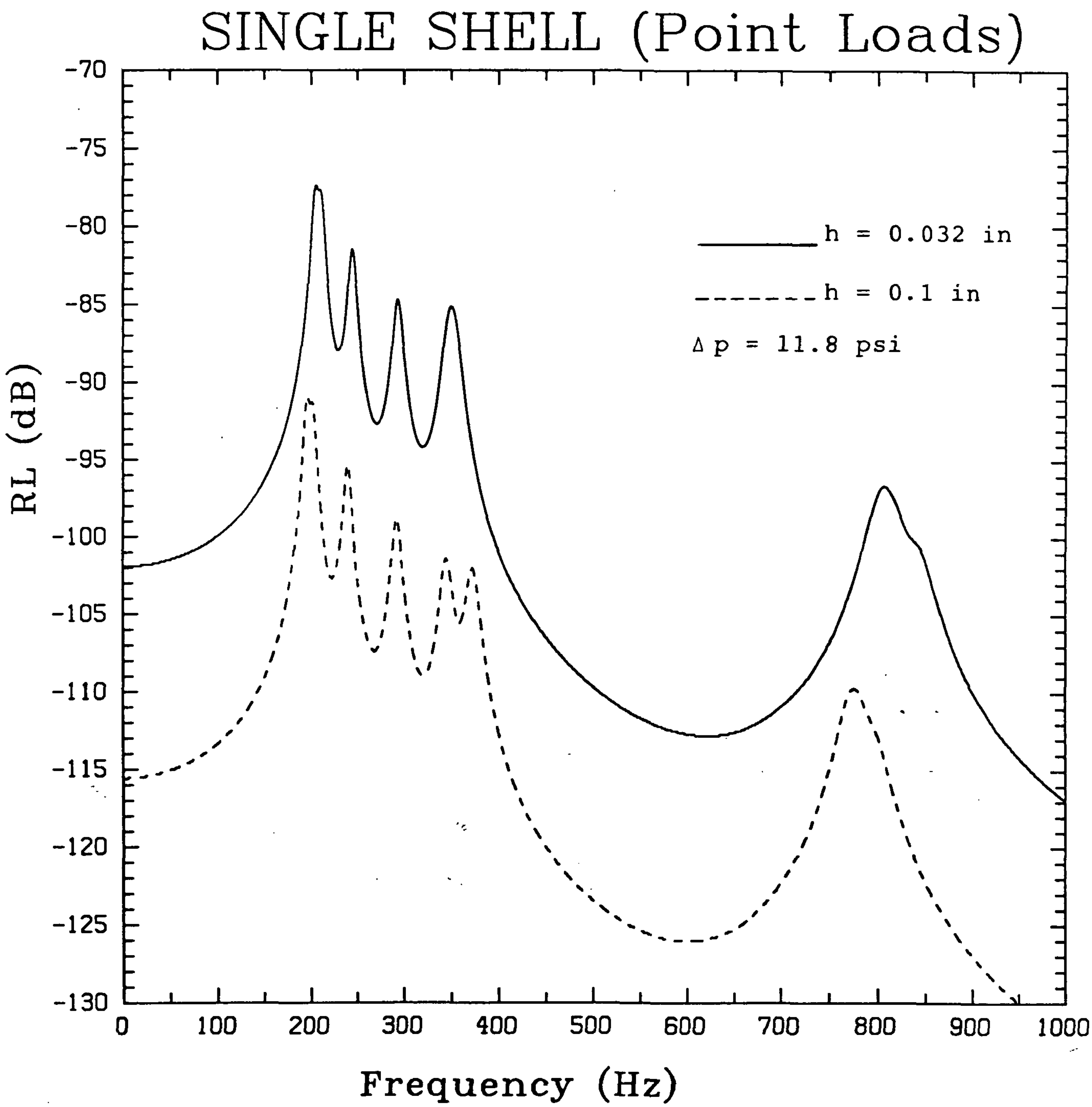

Fig. 6 Relative response levels for a shell stiffened with ten frames (distance between stringers = 10 in)

Frame : 1
File : USERDISK1:[LYRINTZIS.SHELL]FREQ2.OUT;10
Printed : 9-JUN-86 15:43:44

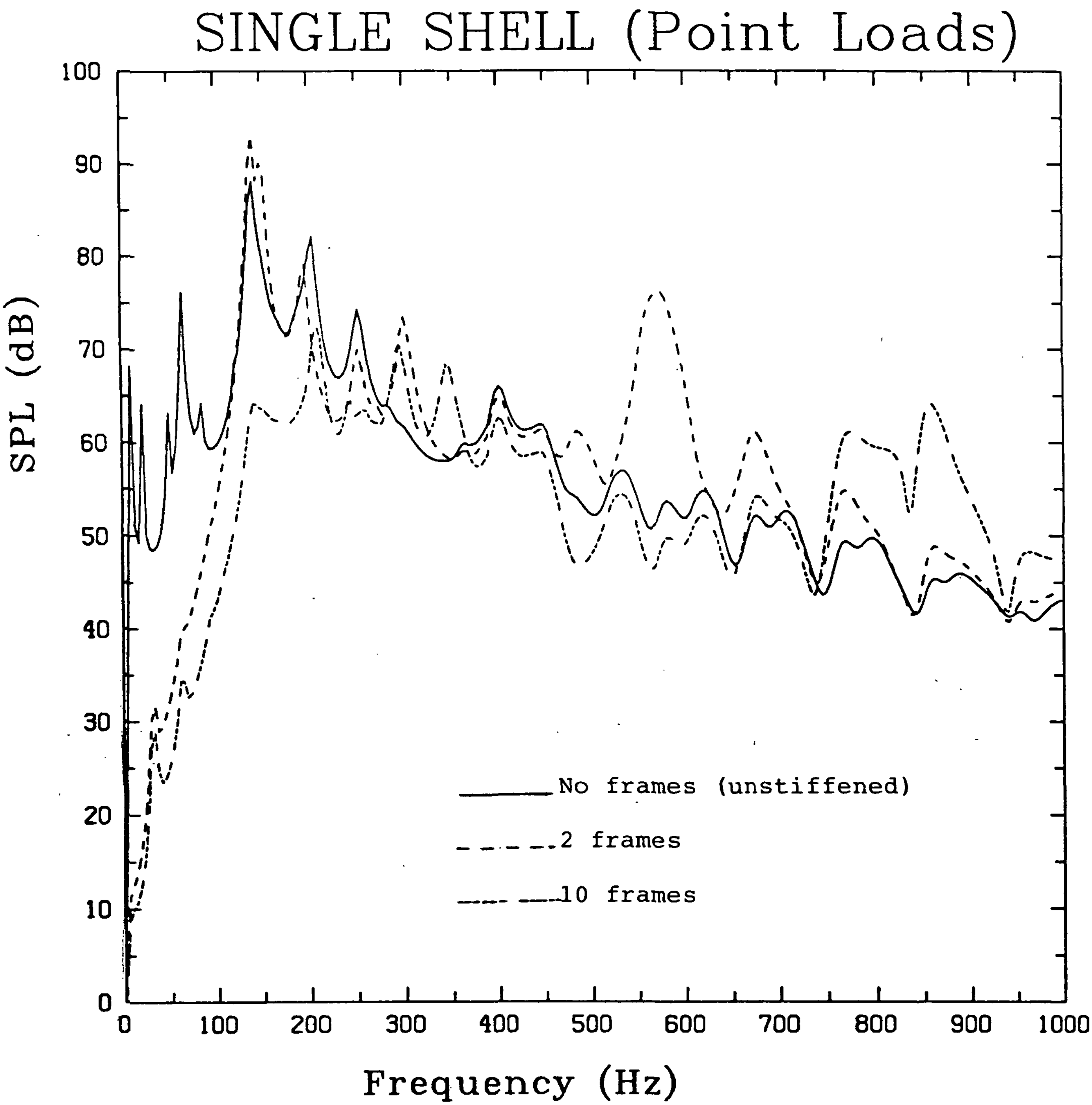

Fig. 7 Interior sound pressure levels for different shell configurations (no stringers)

Frame : 1
File : USERDISK1:[LYRINTZIS]FREQ2.OUT;6
Printed : 27-FEB-86 15:34:58

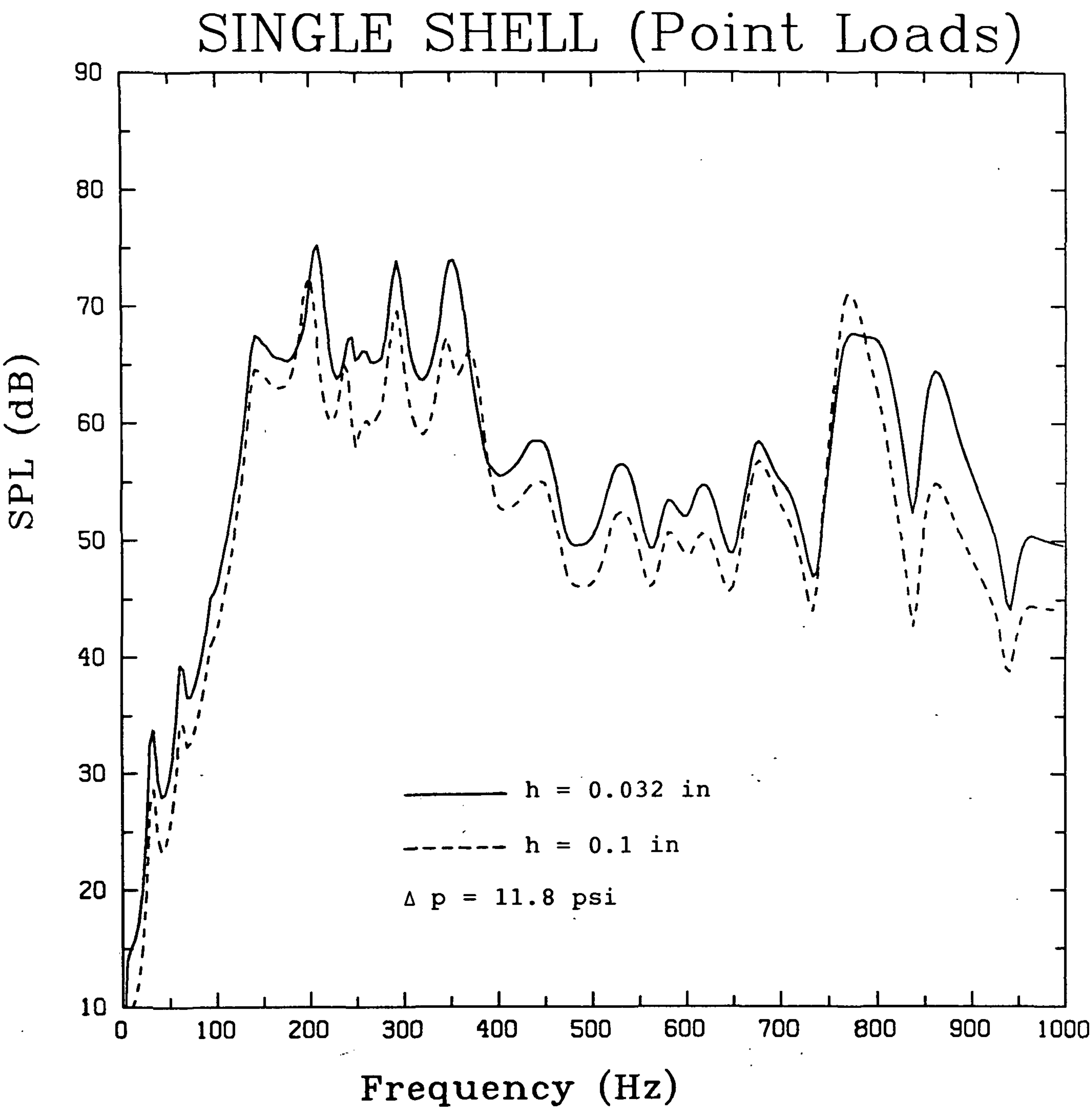

Fig. 8 Interior sound pressure levels for different shell skin thicknesses

Frame : 1
File : USERDISK1:[LYRINTZIS.SHELL]FREQ2.OUT;2
Printed : 9-JUN-86 14:20:56

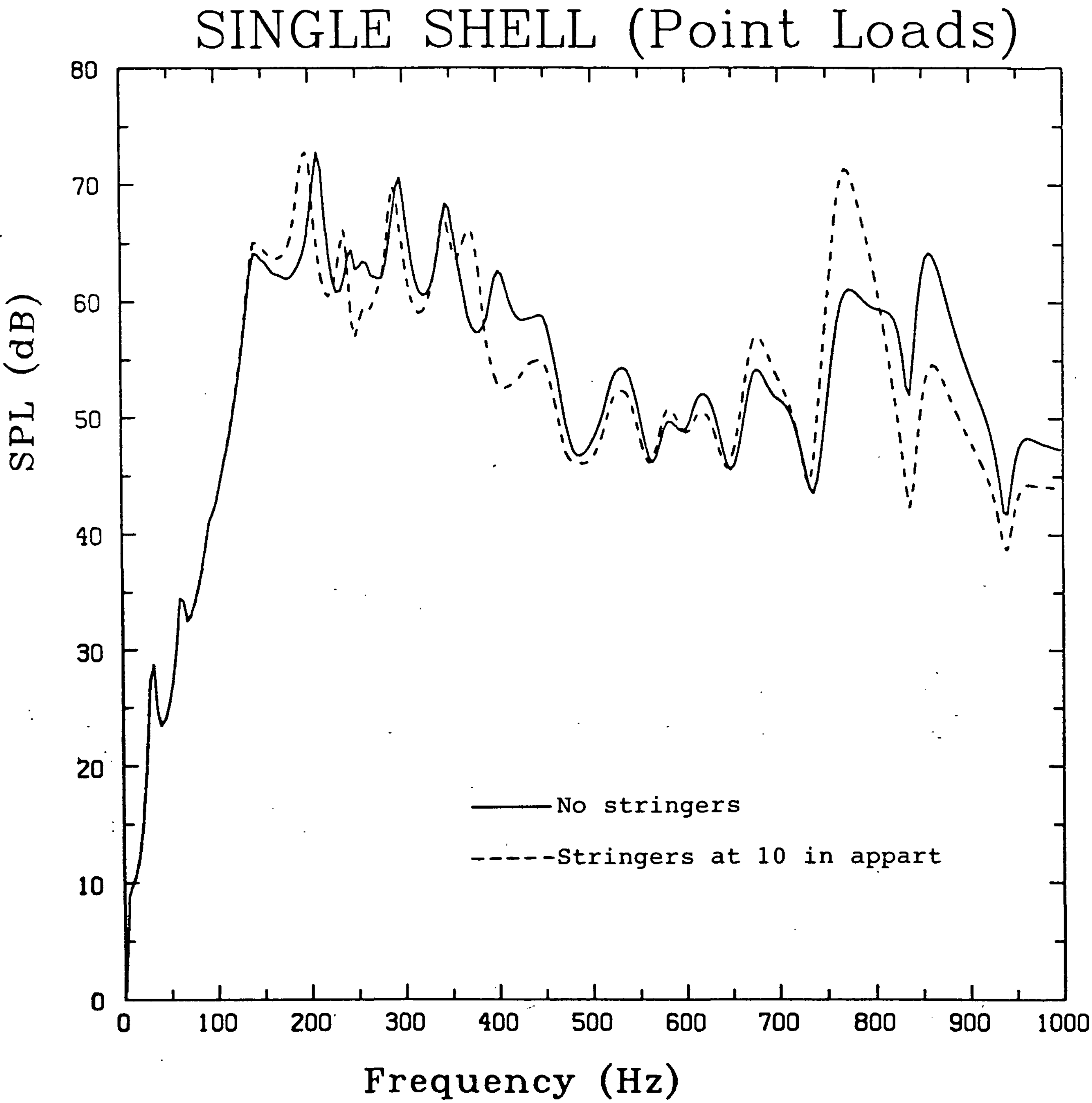

Fig. 9 Interior sound pressure levels for a shell with and without stringers

Frame : 1
File : USERDISK1:[LYRINTZIS]FREQ2.OUT;1
Printed : 27-FEB-86 17:32:48

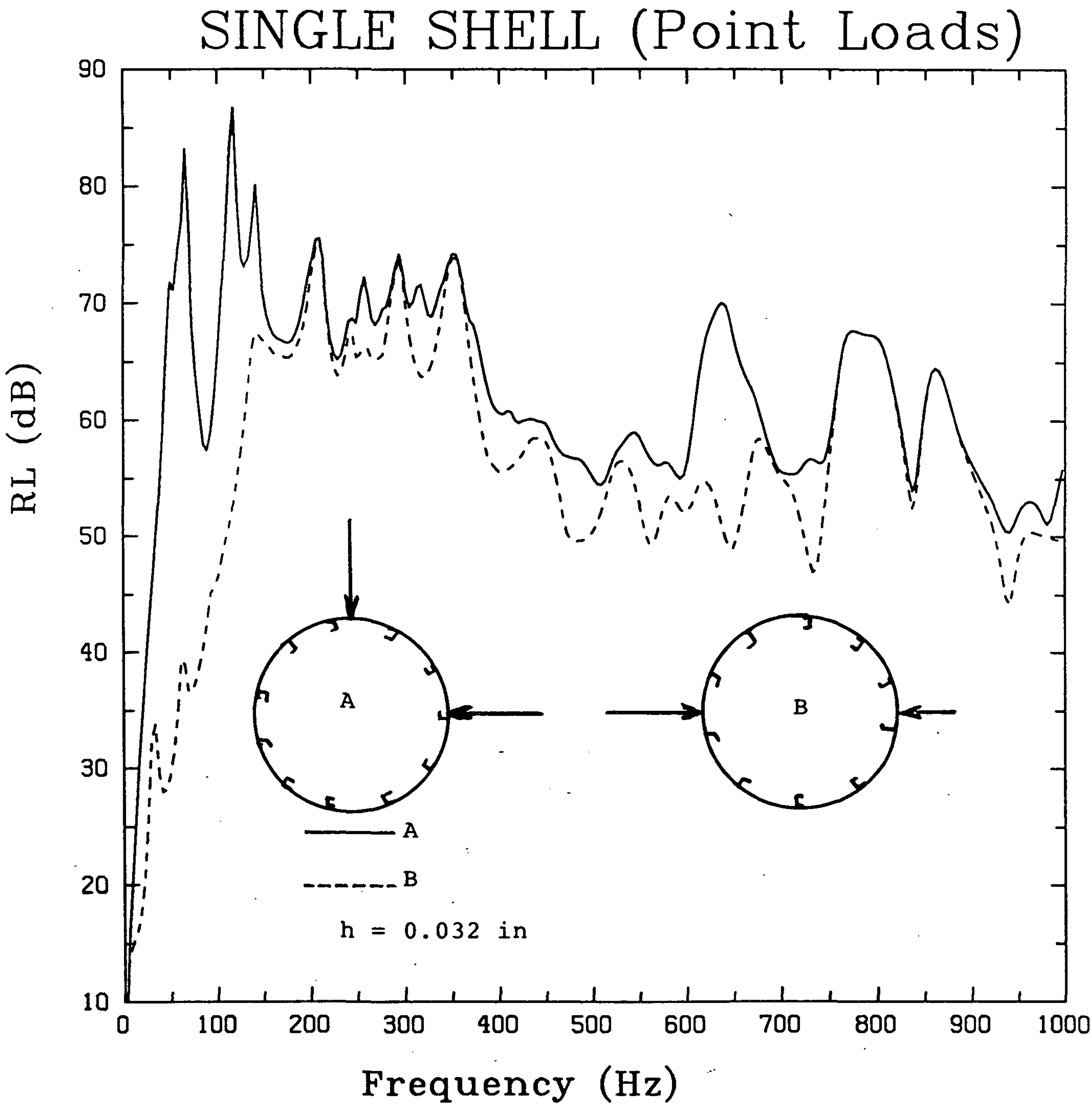

Fig. 10 Sound pressure levels for different locations of input forces

Frame : 1
File : USERDISK1:[LYRINTZIS.SHELL]FREQ2.OUT;1
Printed : 11-JUN-86 11:20:51

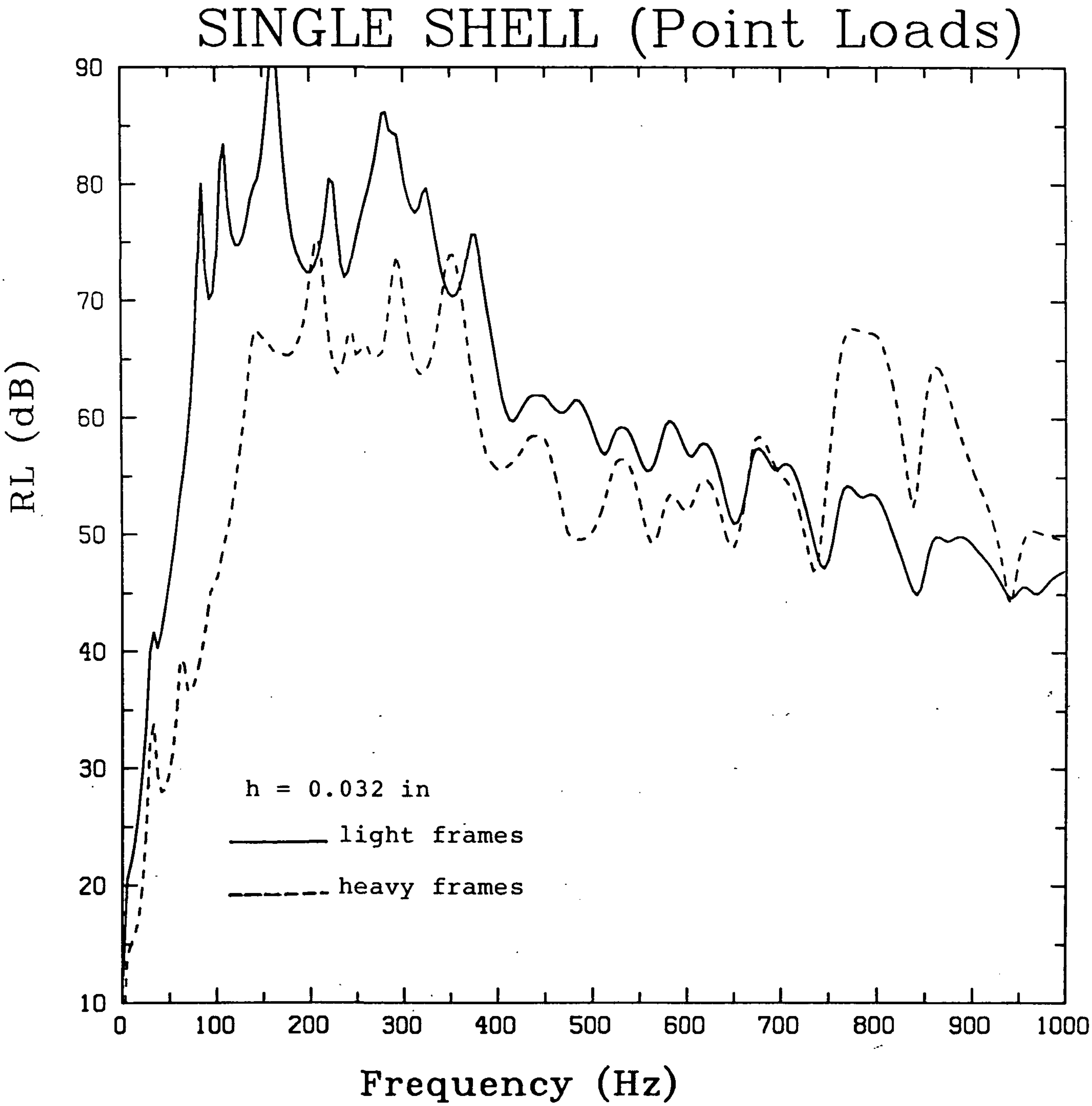

Fig. 11 Sound pressure levels for a cylinder stiffened with heavy and light frames (ten frames)

Frame : 1
File : USERDISK1:[LYRINTZIS.SHELL]FREQ2.OUT;4
Printed : 11-JUN-86 11:42:23

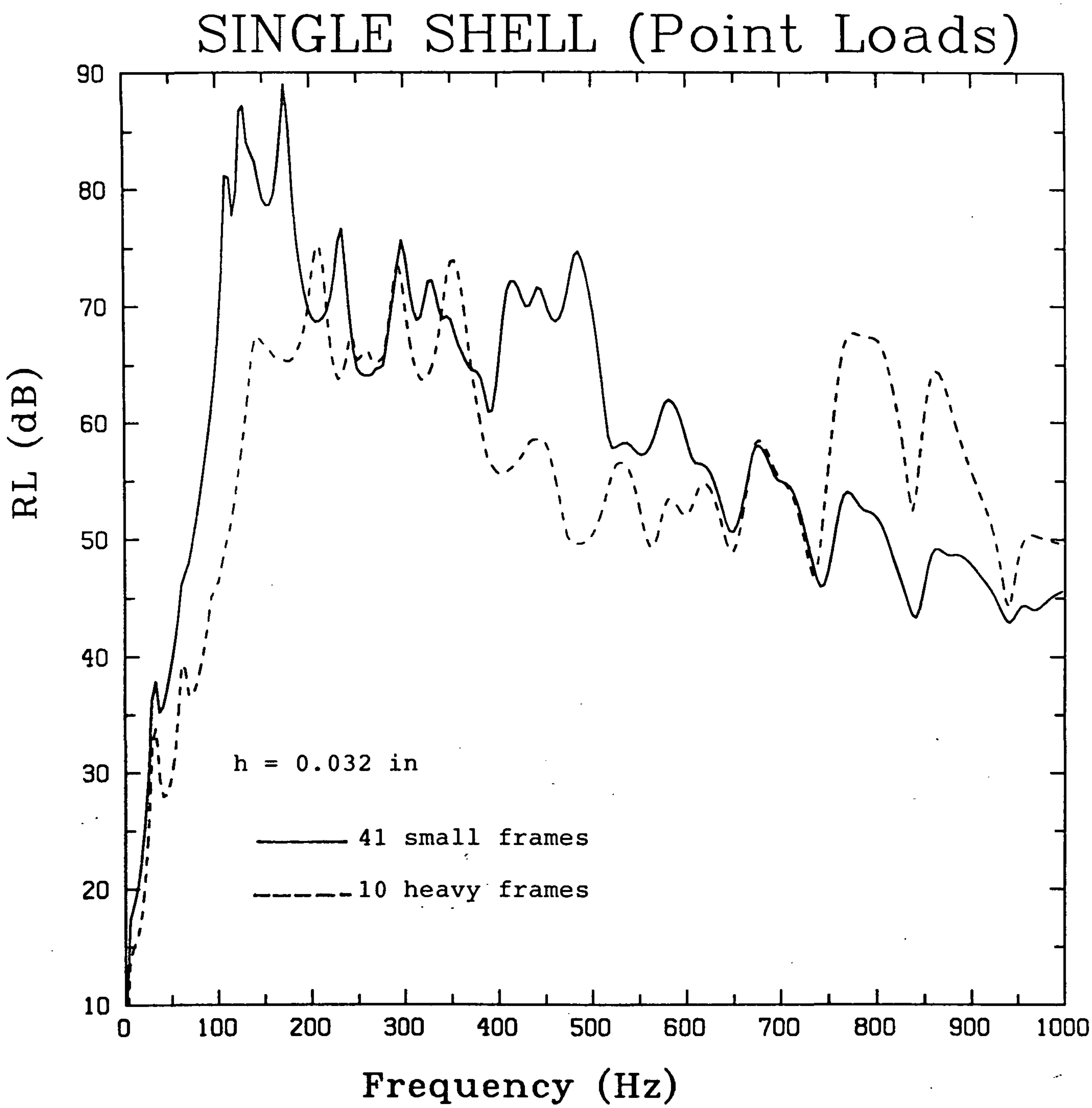

Fig. 12 Interior sound pressure levels for a shell stiffened either with 10 heavy frames or 41 small frames

Frame : 1
File : USERDISK1:[LYRINTZIS.SHELL]FREQ2.OUT;3
Printed : 11-JUN-86 11:38:20

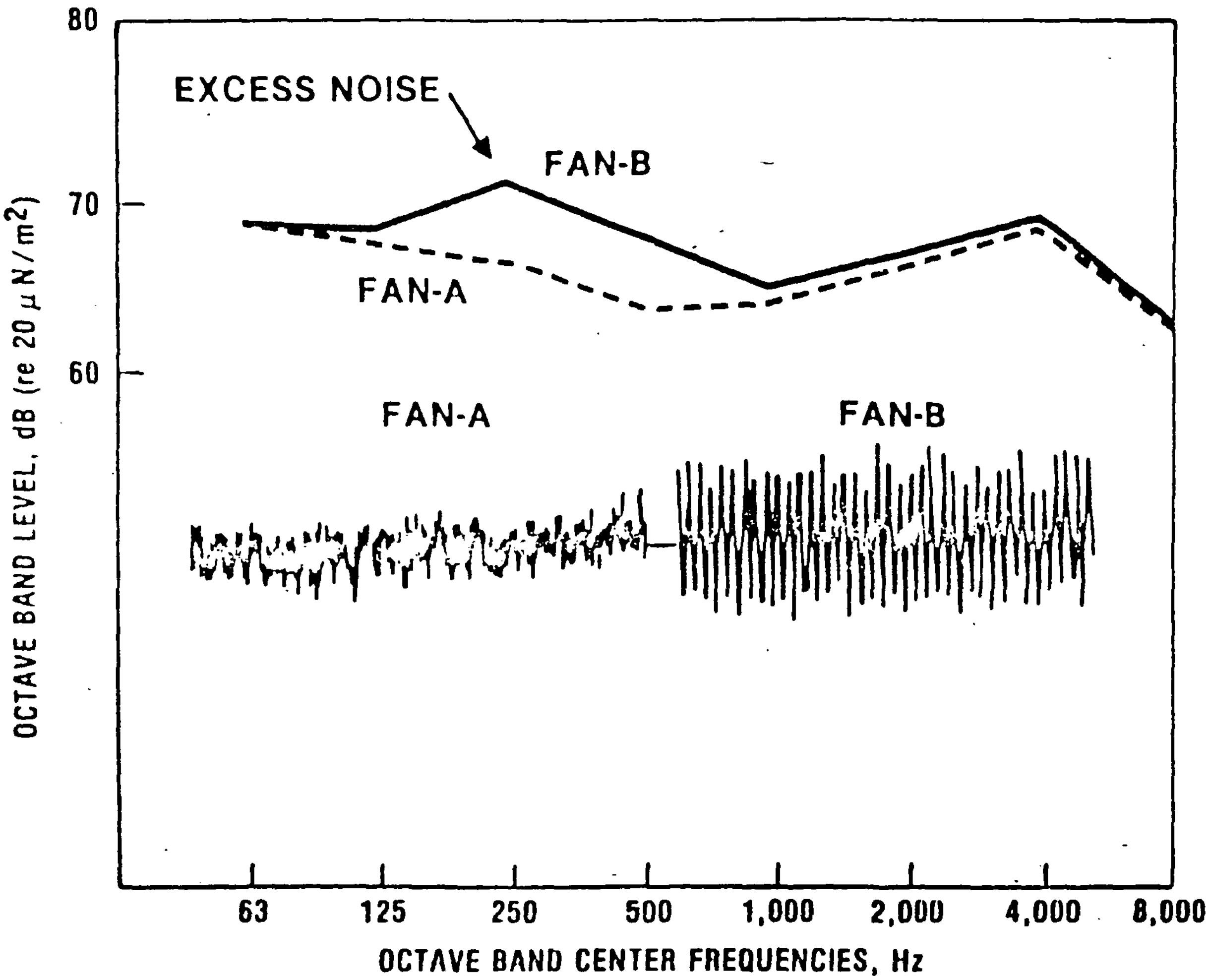

Fig. 13 Interior noise levels generated inside Skylab from operation of two different fans

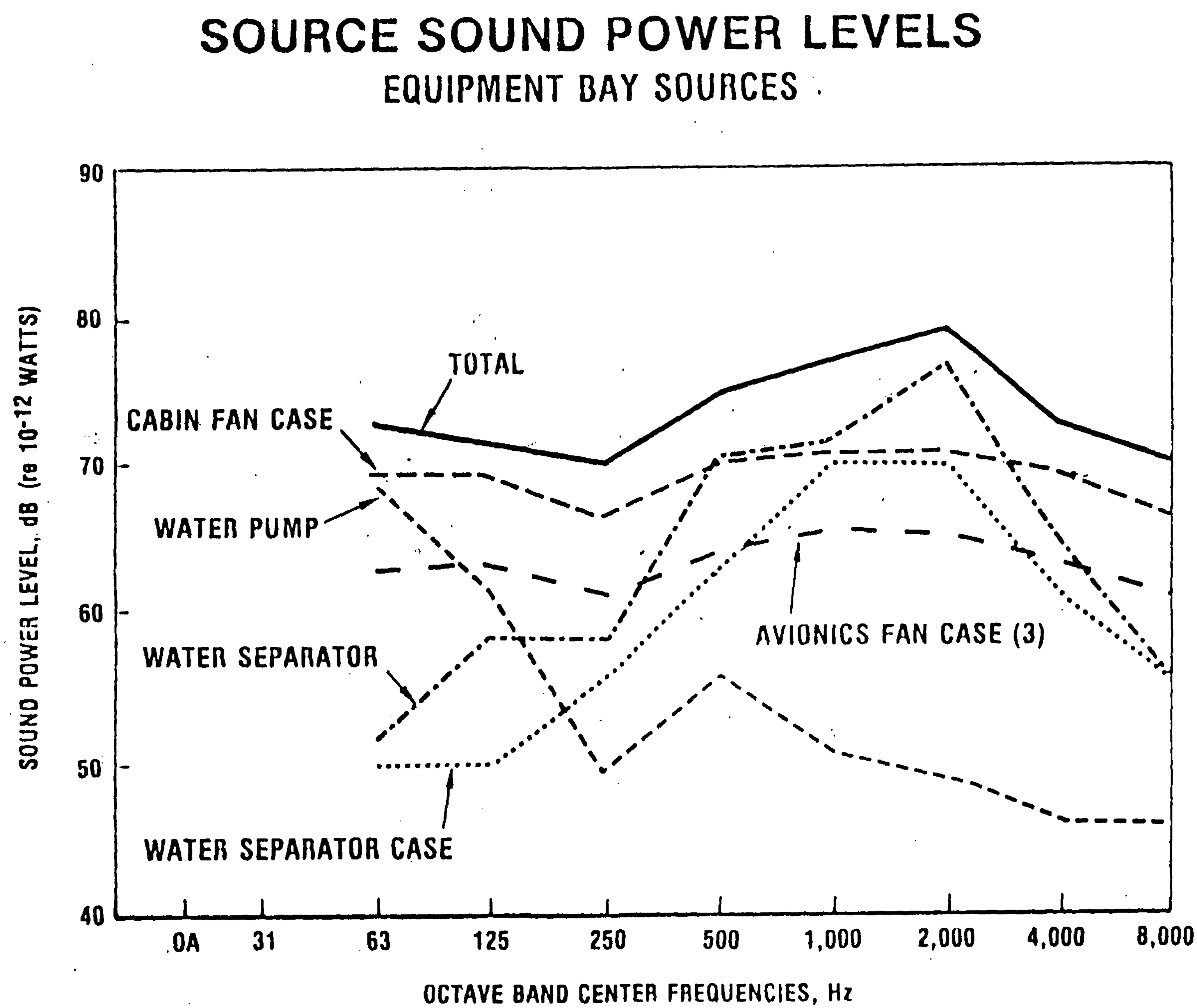

Fig. 14 Interior noise levels inside the Skylab from equipment bay sources